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Hydrodynamic Winch for Salvage Operations

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HYDRODYNAMIC WINCH FOR SALVAGE OPERATIONS

by

E. N. Rosenberg

Ocean Technology Department

October 1972

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NAVAL UNDERSEA CENTER, SAN DIEGO, CA. 92132

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13 ABSTRACT

Lifting heavy loads (1000 tons or more) from ocean depths using the present winching or lifting methods is, for practical purposes, nonexistent. Pontoons and winches have been used for limited loads and limited depths, but breakout forces, buoyancy controls, dynamic-wave induced cable stresses, etc., have made very heavy lifts almost impossible. The hydrodynamic winch is a proposed solution to this problem.

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SUMMARY

PROBLEM

Design a method of salvaging very heavy loads (1000 tons or more) from ocean depths through controlled lifting.

RESULTS

A 3-ft-diameter by 9.5-ft-long working model of a new lifting concept, called a hydrodynamic winch, was constructed and tested under simulated ocean-wave concepts. The model tests indicated that it is possible to control the lifting and lowering of extremely heavy loads at sea using the hydrodynamic winch. The lift device automatically adjusts to the load being lifted and is able to cope with the problem of the additional force that is required for breakaway from the ocean bottom. In addition, the tests showed that the hydrodynamic winch is a stable device that is nonresponsive to normal and relatively high sea states.

CONTENTS

INTRODUCTION

The loss in 1963 of the submarine USS THRESHER (SSN 593) in 8400 ft of water emphasized the need for a method salvaging very heavy loads from ocean depths through controlled lifting.

Pontoons and very large multipart-purchase winch systems have been used for limited loads and depths. The problems of breakout forces, buoyancy controls, and dynamic-wave-induced cable stresses have made the lifting of very heavy loads from ocean depths extremely difficult.

To test the feasibility of a new lifting concept, a 3-ft-diameter by 9.5-ft-long working model of a heavy lift concept, called a hydrodynamic winch (hydro-winch), was constructed and tested under simulated ocean-wave concepts. The results of these model tests indicate the feasibility of the concept.

DISCUSSION

DESCRIPTION OF THE SYSTEM

Figure 1 illustrates the hydrodynamic winch concept which consists of bow, stern, and center sections.

The center section is a cylindrical drum 62 ft in diameter and 177 ft in length. The large cylinder floats in the water and is wound on the outside with large heavy-duty lifting cables. The cables are wound in pairs around the drum. Each cable of a given pair is symmetrical about the longitudinal center of gravity with its corresponding mate. The magnitude of the total load to be lifted determines the size and number of pairs of lifting cables. The winch-lift lines have right- and left-hand leads to maintain symmetry of loading about the longitudinal center of gravity.

The function of the bow and stern sections is to give the hydro-winch a low-drag streamlined shape. These sections, mounted on trunnions, permit the cylindrical drum section to turn without imparting rotation to the bow and stern sections.

Figure 2 is a typical cross-section of the lift-producing center section. The hydro-winch is configured on the inside with a number of radially oriented pie-shaped watertight compartments. These compartments have several transverse bulkheads that are built in to minimize the "free surface effect" of the water. In addition, a manifolding, controlled-water-pumping, and distribution system is included to produce an off-center continuous water-ballast lift torque. The manifolding, water-valving, and sequencing system is illustrated in figure 2 by the passages or tubes that radiate outward from the center of the winch drum.

The sump compartments (a typical cross-section is shown in figure 3) are located adjacent to and between the pie-shaped compartments. A gimbal-mounted submersible pump, which hangs submerged in the sump, is supported by a hollow trunnion located along the longitudinal center of the hydro-winch. The pump hangs from this trunnion and is supported by a vertical hollow tubular support.

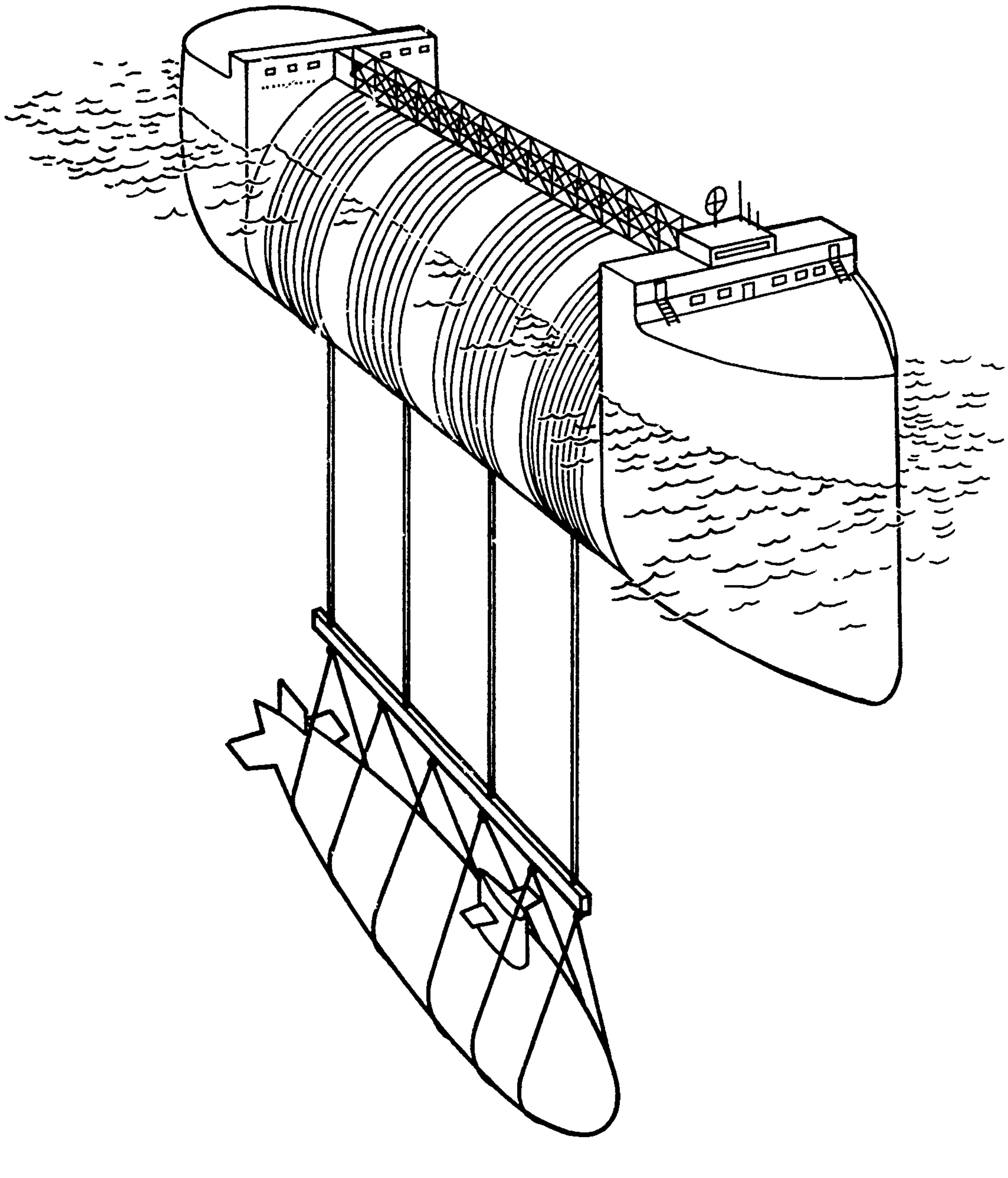

Figure 1. Hydrodynamic winch transporting a salvaged submarine.

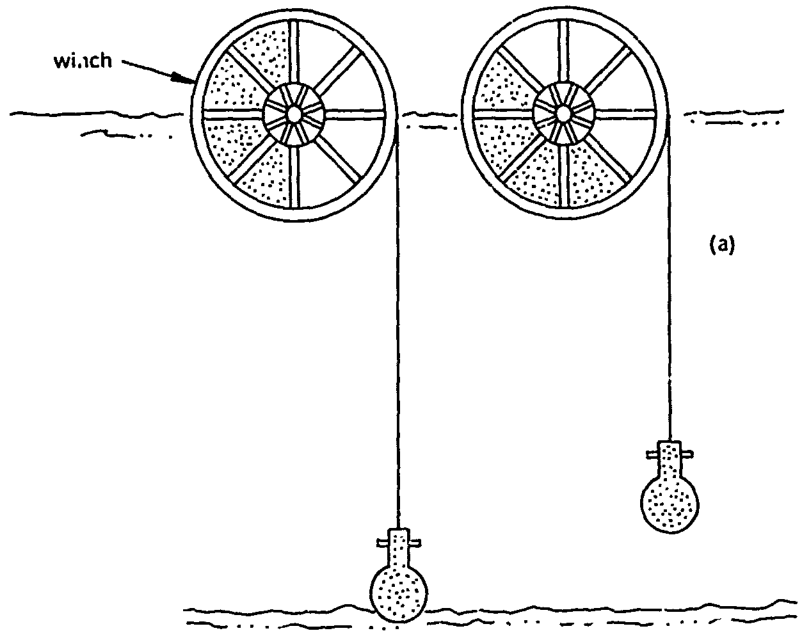

Figure 2. Principle of operation for hydro-winch; note the ballasting of torque-generating compartments during and after breakout of the submarine from the ocean bottom.

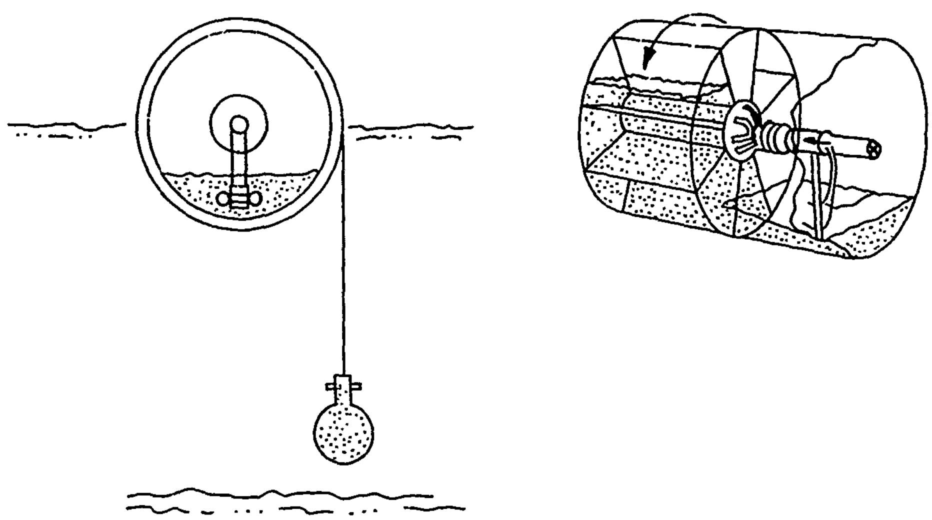

Figure 3. Schematic of sump compartments in hydro-winch.

An inboard view of the winch drum floating in water (figure 4) shows four lift-producing transverse compartments and two sump compartments with gimbal-mounted pumps. The pumps are trunnion mounted and hang from the central-distribution manifolds.

OPERATION OF THE SYSTEM

The hydro-winch drum floats in the water and acts both as a displacement hull and as a winch drum with buoyant rather than mechanical support.

Rather than using an extremely large gearing system to produce the torque, as is typical in conventional winch systems, the torque is produced by continually shifting large amounts of water ballast from the lower sump to the pie-shaped compartments. As the water is pumped, an off-balance torque is produced to rotate the winch drum, thereby lifting the load attached to the lift strongback.

The hydro-winch functions in a manner similar to that of a gearless power-transmission system. The amount of water pumped tends to keep the entire lift system in equilibrium during a lifting or lowering operation. The water pumped from the sump into the upper compartments always balances the load to be lifted. For a relatively light load, a small amount of water is transferred to create the lift, and for a heavy load, large amounts of water are pumped. This is similar to a very large gear reduction for heavy loads and to a small gear reduction for light loads.

All pie-shaped compartments are full on one side of the lift drum (figure 2). In this condition, the winch drum can exert an additional 20 to 30 percent increase in torque or lift for breakout of the load that is embedded in the ocean bottom. (This additional lift would only be used to produce a breakout force.)

Once breakout has beem accomplished, the winch drum approximately orients itself as indicated in figure 2(a). The entire winch system with its load settles to an equilibrium condition. In this condition, the winch and its load have a righting moment and can resist forces to rotate it from its position of equilibrium. This righting moment produces a safety factor to prevent overrunning of the load in heavy seas.

To produce a further lifting of the load, the water is drained from the lower right-hand compartment into the sump and then pumped to the upper left-hand compartment, thereby creating an additional lifting torque.

As the winch drum rotates, the gimbal-mounted pump hangs vertically in the sump, enabling water to be pumped from the sump to the upper compartments. The water is pumped up through the vertical pump-supporting tube through the gimbal or trunnion into the sequencing and compartment-distribution system.

As the winch drum rotates, the water is valved and manifolded to drain into the sump from the lower pie-shaped compartments. The water, continually circulating during a lifting operation, is supplied by the pump which continually shifts large amounts of ballast.

The magnitude of the lift is limited by the displacement of the lift device. The speed of the lift is limited by the size of the pumping system. A lift of 1000 tons at 2.5 ft/min will require a total pump capacity of 6000 gal/min. A small pump can lift an extremely heavy load if time is not important.

For lowering a load, the pump is not needed. If water is allowed to flow from one pie-shaped compartment to another around the periphery of the winch, the load lowers. The speed of lowering is determined by the rate of water flow.

To stop the winch under load in midwater, the pumps are turned off and no water is allowed to flow from compartment to compartment.

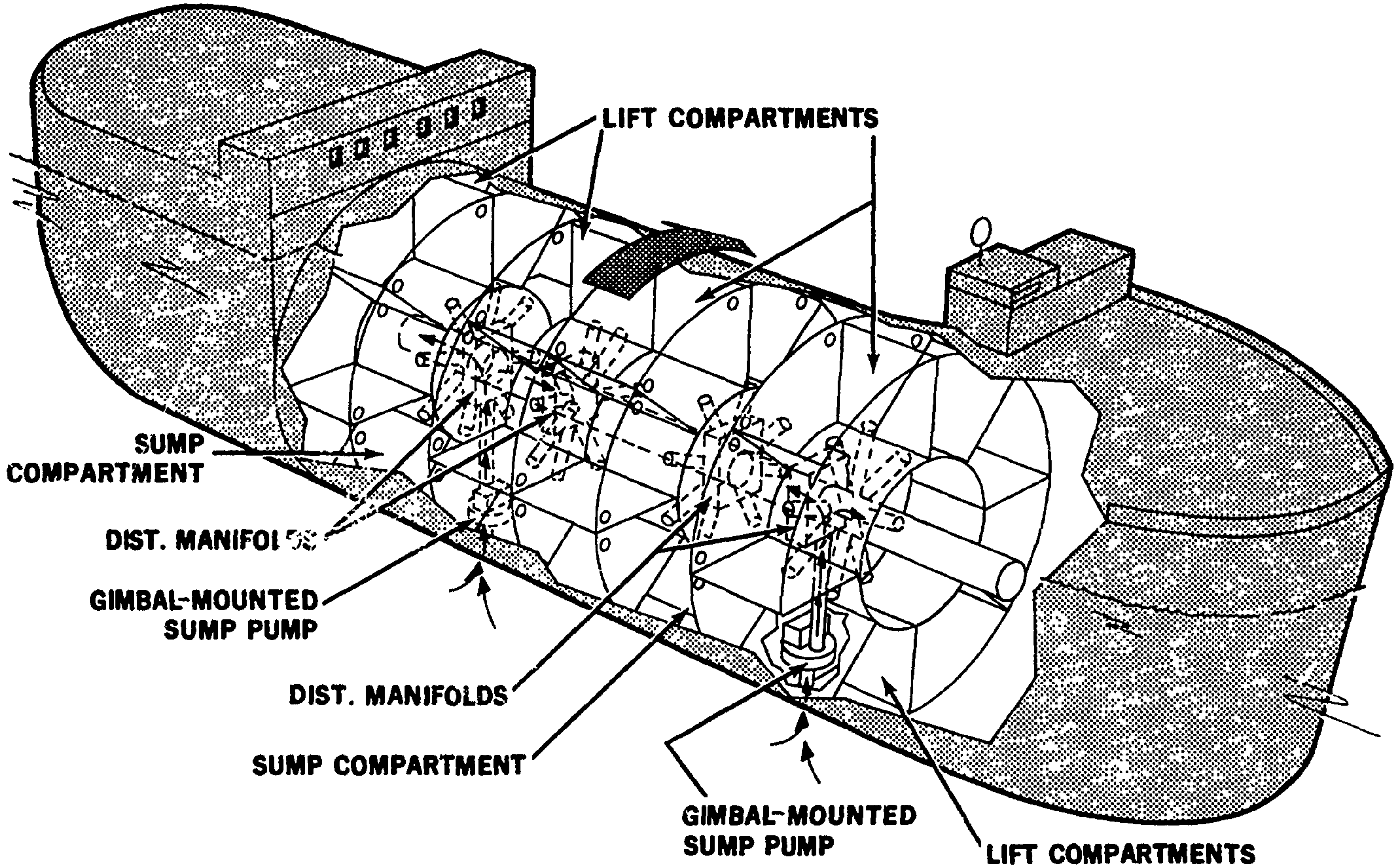

Figure 4. Major compartments of hydro-winch system.

The entire lifting or lowering operation is very simple, and the load is under control at all times. The rate of lifting or lowering can be controlled, as well as the magnitude of force applied during breakout of the load from the ocean bottom.

During breakout, the winch turns approximately 40 deg and then settles to equilibrium about 20 deg from the original position with the load suspended from the bottom. The load is under control, and it can be brought to the surface by continuation of the pumping.

For lowering, the large lifting strongback is sufficiently heavy to create a downward pull and thus unreel the lift cables off the drum for attachment to the load to be lifted.

MODELS

DESCRIPTION

A 35.5-in.-diameter by 112-in.-long working model of the hydro-winch was constructed and tested. The winch used two off-the-shelf 1/3-hp submersible sump pumps for transferring the water ballasts. The sequencing valves for draining and filling the compartments were pneumatically operated using small pilot valves operating from an adjustable sequencing cam.

Figure 5 shows a model that is designed to lift 400 lb with the bow and stern in place. Figure 6 shows the drum section floating in the water-ballasted low for the lift condition. The control panel, the interconnecting pneumatic hoses, and electric power lead can be seen in the foreground. Figure 7 is an underwater view of the winch drum with a simulated model-submarine load being lifted. Two lift lines, the strongback and lifting slings, are visible.

The model was first tested in place on the special trailer shown in figure 5. The model was partially filled with water, and the controls and pumps were operated to ensure that the winch functioned properly. In these tests, the bow and stern sections were omitted on the model. The transverse roller wheels on the trailer allowed the winch drum to rotate, providing an opportunity to adjust the valve sequencing cams.

After the proper adjustments were made, the hydro-winch was placed in water and sequentially operated in the lift and lowering modes.

MODEL TESTS

Starting, lifting, and stopping of the lift operation were very smooth in still water. The lift model acted like a variable-displa[illegible] hydraulic transmission: the greater the load, the slower the lift; the lighter the load, [illegible] the lift.

Breakaway tests were also made by placing an excess, or breakaway, weight on the model. This weight simulated the extra load encountered when pulling an object out of mud. The total load lifted by the model was 405 lb, but approximately 150 lb of this load were suddenly released when the load broke free of the bottom. The only effect was that the lift drum rotated about 40 deg upon release of the weights and came to rest about 20 deg from the original lift position.

The model was tested in the General Dynamics Wave Generating and Towing Basin, San Diego, California, for simulated sea conditions, including critical periods of waves for roll, pitch, and heave. The towing basin is 300-ft long by 12-ft wide by 6-ft deep. During

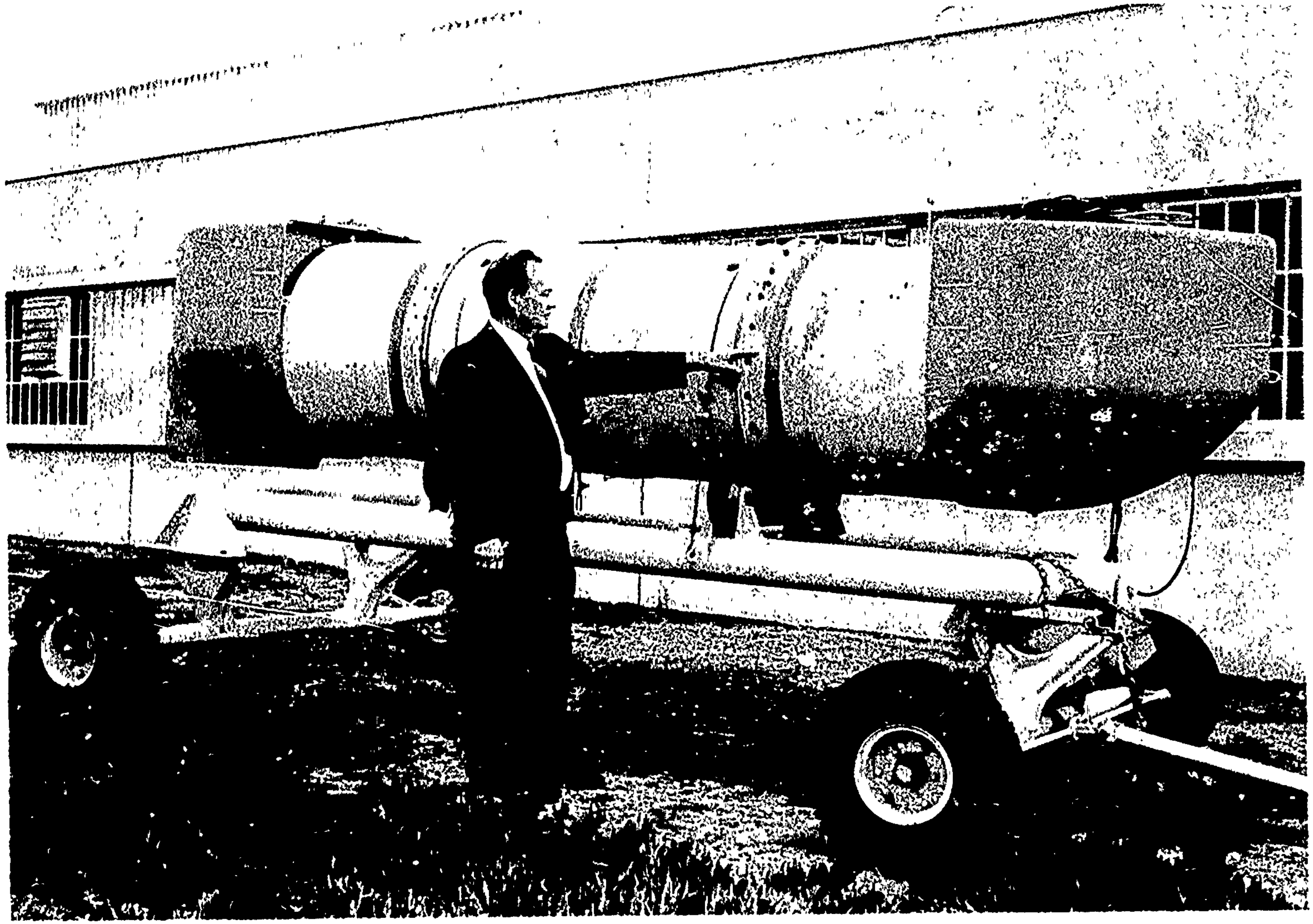

Figure 5. Hydro-winch model with 400-lb lift capacity.

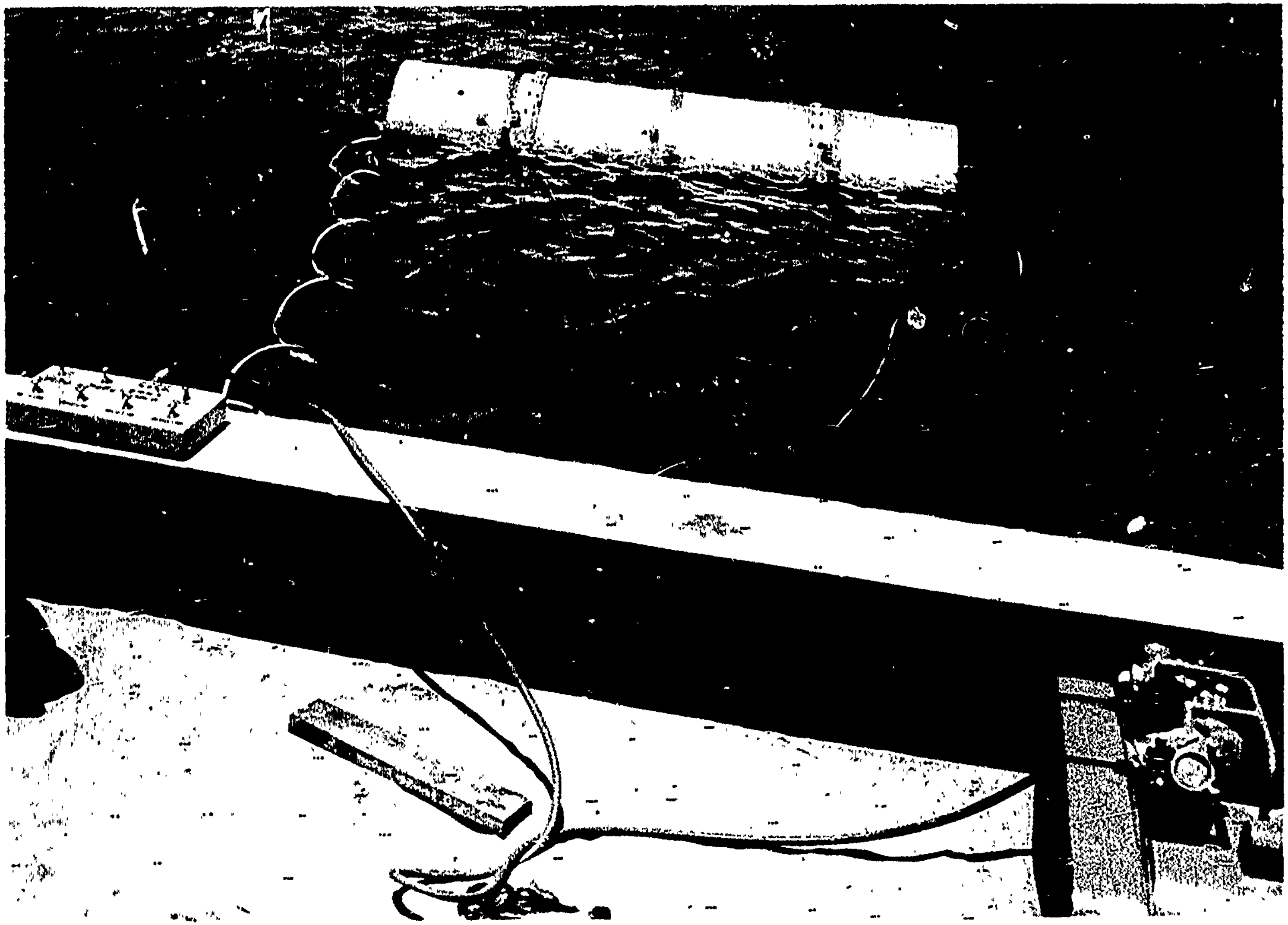

Figure 6. Drum section from the hydro-winch model during actual lifting tests.

Figure 7. Underwater view of the hydro-winch drum section and the 400-lb disabled submarine model during simulated recovery operations.

the tests, the waves had very little effect on the model winch. The only effects seemed to occur at the critical wave periods when the maximum observed roll was about 10 deg off each side of center.

Strain gauges were used in the lift lines. The observed increase in load resulting from the acceleration forces was only 10 percent of the static load. Apparently, the 2000-lb mass of the lift drum, when compared with the 405-lb lifted load, minimized the wave's effects.

The lift device also worked well when tested using a counterweight to increase the total lift capacity. It is anticipated that an almost double lift capacity would be realized using this technique if the total displacement of the hydro-winch could provide the needed buoyancy for both the load being lifted and the counterweight.

The device operated well in the lowering mode with both pumps turned off, allowing the water to circulate sequentially from the higher compartments to the lower compartments. Both the lift and the lowering functions proved to be smooth and controllable. Starting or stopping did not produce any noted oscillations or over-travel. The load could be stopped in midlift and would remain stable for normal operations. The critical periods of heave and pitch varied for the model, depending on load and ballast conditions. The period ranged from approximately 2 sec for heave and pitch to 5 sec for roll.

Using appropriate scaling factors, a full-size 1000-ton winch would have a roll period of approximately 22.4 sec. A winch capable of lifting 5000 tons would have a roll period of approximately 30 sec, and a 10,000-ton lift device would have a roll period of 33 sec. These long wave periods are beyond those expected for normal sea states.

TOW TESTS

After completion of the lifting and wave-response tests, a fiberglass bow and stern were trunnion mounted to the ends of the winch drum. A series of tow tests was run in several simulated sea conditions. These tests were conducted at two conditions (63.4 percent draft and 81.0 percent draft) over a range of forward velocities from 0.77 ft/sec representing 2.07 knots to 10.2-knot speeds for a 1000-ton lifting-capacity hydro-winch. The model was bridle-towed in smooth water and in waves 4.7-in. high with periods of 1.5, 2.0, and 2.5 sec. These wave heights and periods represented actual 8.06-ft-high waves at sea and 6.84-, 9.1-, and 11.4-sec periods. The average towing resistances were measured using a small force dynamometer incorporated into the lines, and the resulting measurements were converted to a coefficient form.

At the 63.4 percent draft condition, the addition of the bow and stern sections reduced the drag coefficient from $C_D = 1.60$ for the plain cylindrical hull to $C_D = 0.18$ for the ship-shaped hull.*

The model hydro-winch was also tested in the lift and lowering mode with the bow and stern of the model maintaining upright positions while the center portion or the winch drum rotated. This was accomplished by proper ballasting of the bow and stern to produce an adequate righting moment to resist the frictional turning moments that were produced by the rotating drum. With the bow and stern sections locked in place for foul-weather conditions, an additional righting moment will occur from the shift in buoyancy of the bow and stern sections. In addition, it is anticipated that by proper proportions, heave and pitch can be minimized.

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*Ball, Steve. *Stability and Resistance Model Tests of the Hydrodynamic Winch Deep Submergence Salvage Vessel.* General Dynamics Electric Boat Division, Report No. U419-66-011, Groton, Conn., July 1966.

RESULTS

The model tests indicate that it is possible to control the lifting and lowering of extremely heavy loads at sea using the hydro-winch. The lift device automatically (within its capabilities) adjusts to the load being lifted and is able to cope with the problem of the additional force that is required for breakaway from the ocean bottom.

The hydro-winch tends to be a stable device that is nonresponsive to normal and relatively high sea states. The large mass of water in the lift device produces relatively long heave, roll, and pitch periods. The many compartments filled or partially filled with water reduce the free surface effect (sloshing) to a minimum, producing a hull that is almost nonresponsive to wave motions and forces. The cylindrical outer cross-section of the winch hull in the flooded low position is also almost nonresponsive to waves. A very minimal heeling moment is generated by a wave moving across the transverse hull section; however, the heeling moment is insufficient to produce an appreciable roll in the hydro-winch.

For all conditions of lifting or lowering, the buoyant forces, payload, and weight of the lift drum and contained water act together along a vertical line passing through the lift drum's center of buoyancy because of the structural symmetry in the system. This creates a lift system that is always stable within its lift capability, and thus eliminates any rolling or unrolling tendency of the lift lines on the lift drum.

The model performed well under all simulated test conditions for breakaway as well as for various sea states, emphasizing the feasibility of a full-size device.

CONCLUSIONS

The hydro-winch provides a simple yet effective method of lifting, lowering, and controlling very heavy loads at any ocean depth.

APPENDIX A. AUXILIARY HYDRO-WINCH EQUIPMENT

As a self-contained heavy-lift system, the hydro-winch can function as a towed or self-powered salvage and ocean engineering vessel.

To fully utilize the potential of such a large lift system, several auxiliary subsystems have to be devised. Figure A1 shows a subsystem proposed to operate in conjunction with the hydro-winch. The main component of the Deep Ocean Work Station (DOWS) is the strongback to which a suite of underwater tools is attached. The overall configuration of the strongback is similar to a large "saw horse." The legs of the strongback can be adjusted in height by using large salt-water hydraulic rams. Four steel lift cables, fastened to the top of the strongback, are the main lifting lines that are attached to the hydro-winch drum at the surface of the ocean. If necessary, the lift lines can be slackened to decouple the DOWS from the pitch and heave of the hydro-winch on the ocean surface.

The central strongback has two independent traveling cantilever booms. Each traveling boom has an independent traveling manipulator and winch. The manipulators and winches can be moved back and forth along the length of the booms, which, in turn, move along the length of the strongback.

Figure A1 shows a semicircular device partially embracing the small submarine. This device, held in position by the two manipulators, is a track-mounted circling and jetting unit that pulls and lifts cables around and under the submarine hull. A wire rope is attached to the forward end of the jetting unit and is pulled by it around the submarine hull. In this manner, a series of wire rope slings can be sequentially jetted around the submarine hull. These slings, through the use of the manipulators and winches, can subsequently be attached to the strongback above the submarine for lifting by the hydro-winch.

Other work tools that can be attached to the strongback consist of (1) several large "ice tong" grabbers that clamp around a hull or pipe that is to be lifted and (2) a clam shell which is useful for ocean-bottom excavation or mining. The rigid strongback with its high supporting legs and movable booms will, in addition, provide excellent locations for underwater lights illuminating ocean-bottom work areas. Because the lights can be remotely moved at will during the work operations, the backscatter of light will be minimized.

An additional use for the hydro-winch is illustrated by figure A2 where the line pull developed by the winch is used for grounded-ship recovery.

Each DOWS component can be controlled with either a hard wire or an acoustic link terminating at the hydro-winch or a small manned submersible. In either case, the operator will visually monitor the operation of the DOWS suite of tools. If the operator is located in the pilot house of the hydro-winch, he will monitor the work area under DOWS through a closed-circuit television system. However, if the operator is located in the submersible, he will observe the operation of DOWS through a panoramic-vision window.

The power to DOWS will be transmitted by an umbilical cable carrying 3-phase AC current. The umbilical cable, like the lifting lines, will be stored on the hydro-winch drum. When the DOWS is lowered to the ocean bottom, the umbilicals will unreel with the lifting lines.

Figure A3 is a curve plotted to show the lift capacity of a hydro-winch relative to its size.

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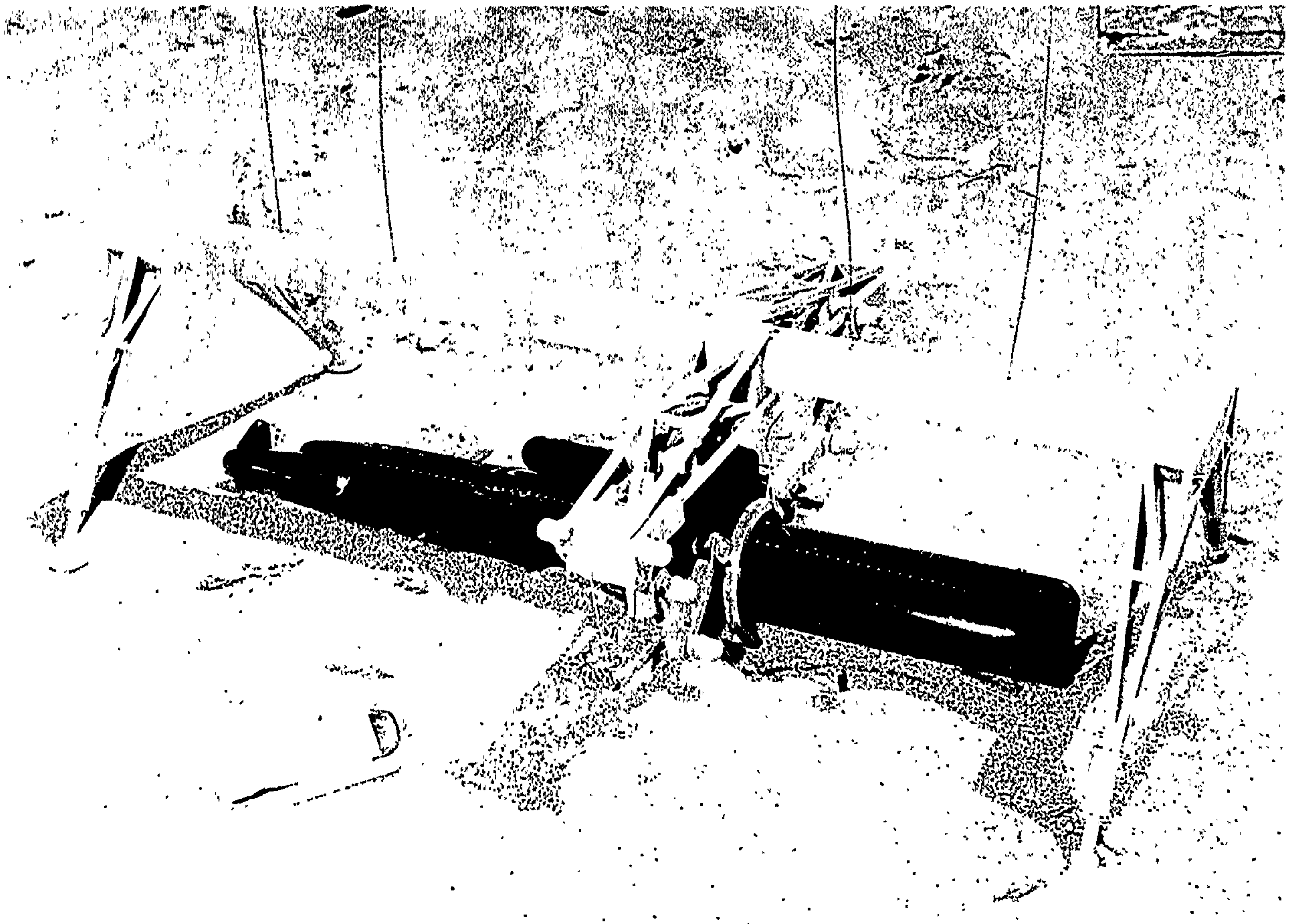


Figure A1. Concept of the hydro-winch strongback that also serves as the Deep Ocean Work Station for attachment of lifting lines to disabled submarines.

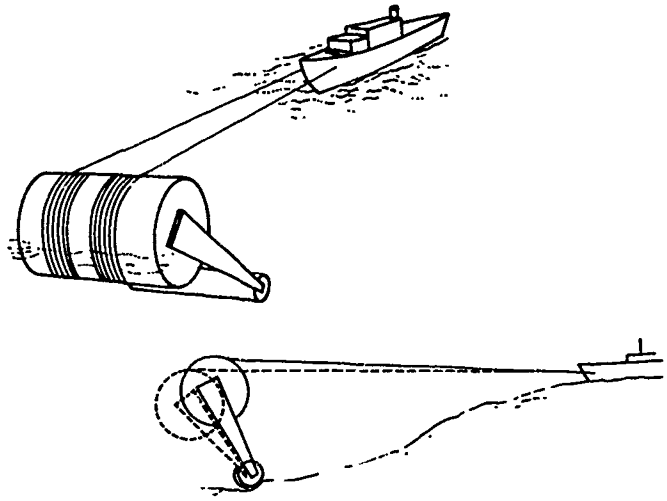


Figure A2. Modified hydro-winch for recovery of grounded ships.

Table A1 gives the approximate wire rope criteria for several sizes of hydro-winches.

Table A1. Wire Rope Criteria.

Parameter	Value				
Design live load, short tons	10	100	1000	5000	10,000
Wire rope criteria					
Breaking strength per line, short tons	6.3	63	630	3150	6300
Breaking strength for four lines, short tons	25.2	252	2520	12,600	25,200
Wire rope diameter, in.	0.43	1.35	4.25	9.50	13.45
Wire rope weight per line, lb/ft	0.3	3.0	30.5	152.5	305
Total weight of line and fittings at 6000-ft depth, short tons	3.84	38.4	384	1920	3840
Total weight of line, fittings, and live load at 6000-ft depth, short tons	13.84	138.4	1384	6920	13,840
Breaking strength/live load	2.5	2.5	2.5	2.5	2.5
Breaking strength/total weight of line, fittings, and live load	1.8	1.8	1.8	1.8	1.8

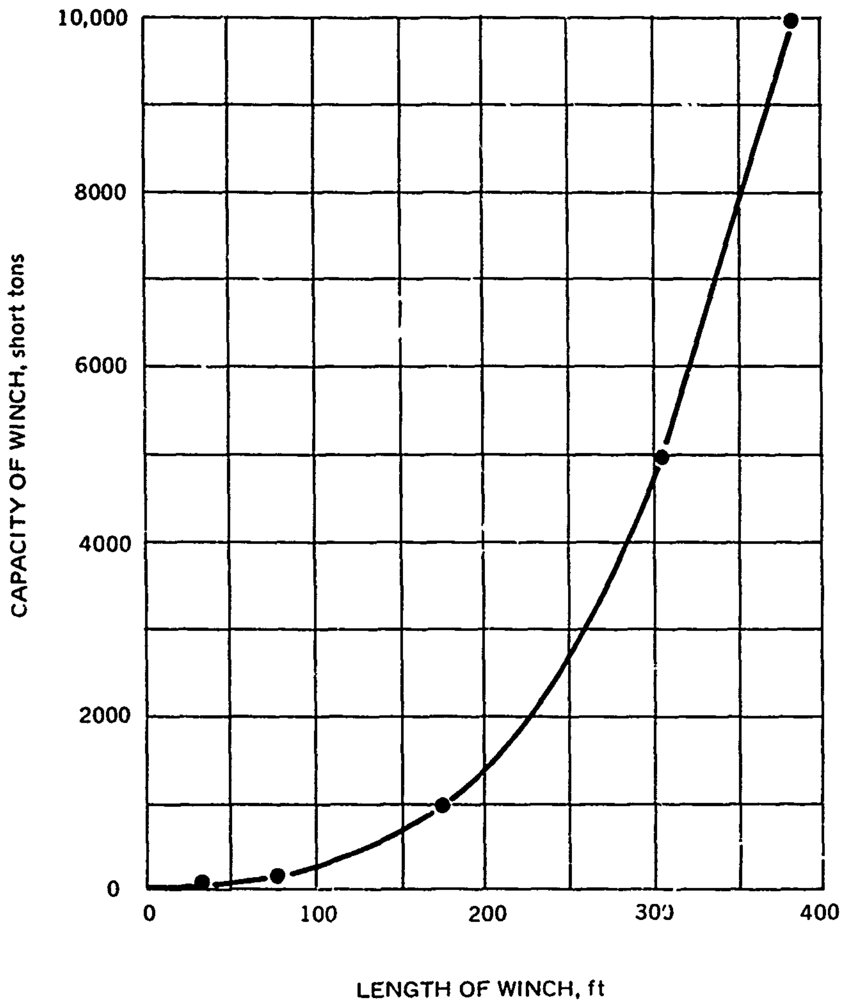


Figure A3. Lift capacity of hydro-winch as a function of the drum length. (The outer diameter is approximately one-third the winch-drum length, and the inner tunnel diameter is approximately one-third the outer diameter.)

APPENDIX B. CALCULATIONS

DISPLACEMENT CALCULATIONS FOR A 1000-TON HYDRO-WINCH

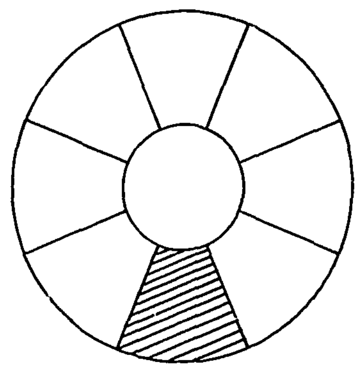
For a 1000-ton lift device (6000-ft depth) the cylinder is 62 ft in diameter and 177 ft in length. The following data also apply.

Estimated weight = 2810 short tons (empty).

Estimated displacement $= \dfrac{\pi \times 31^2 \times 177 \times 64}{2000} =$ 17,100 short tons.

Empty weight = 16.5 percent of displacement.

Draft = 14 ft.



Assume an effective total length of 120 ft for the pie-shaped lifting-producing compartments and fill one compartment (towing condition).

Weight = 2810 ÷ 1250 = 4060 tons.

$\dfrac{4060}{17{,}100}$ = 23.8 percent of displacement.

Draft = 18 ft.

LOAD CONDITION "A"

For a 500-ton lift device at a 6000-ft depth (using two lift cables), the following apply.

Water in compartments = 5000 + 1250	= 6250 tons
Load + cable load	= 710 tons
Weight of structure	= 2810 tons
	9770 tons.

Buoyancy = 17,100 − 9770 = 7330 tons.

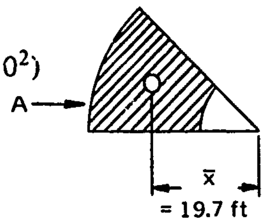
$\dfrac{9770}{17{,}000}$ = 57.2 percent of displacement.

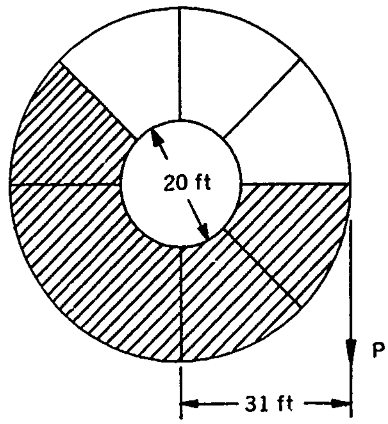
Draft = 35 ft.

Load lift = 500 tons.

Equivalent torque.

One 45-deg sector.





$$A = \frac{\pi}{8}(30.5^2 - 10^2)$$

$$= 365 - 39$$

$$= 326 \text{ ft}^2.$$

$$CG = \bar{x} = \frac{0.6\,[(365)(30.5)-(39)(10)]}{326}$$

$$\bar{x} = \frac{0.6\,(10{,}730)}{326} = 19.7 \text{ ft.}$$

$$T_f = \frac{326 \times 19.7 \times 64}{2000} = 203 \text{ ft/ton/ft.}$$

Total cable pull at drum = 710 tons.

Total lift at 6000-ft depth = 500+ tons.

LOAD CONDITION "B"

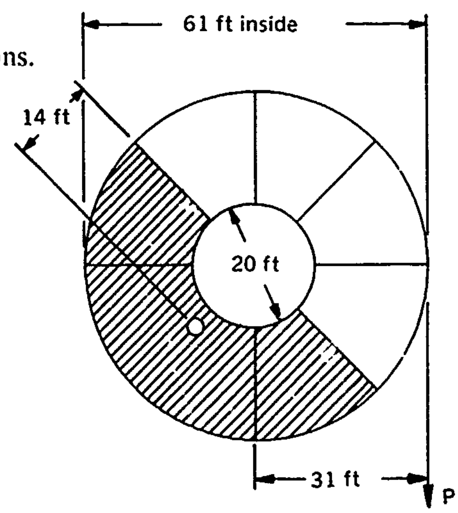
For a 1000-ton lift device at a 6000-ft depth (using four lift cables), the following apply.

Water in compartments	=	5000 tons
Load + cable load	=	1400 tons
Weight of structure	=	2810 tons
		9210 tons.

Buoyancy = 17,100 − 9210 = 7890 tons.

$$\frac{9210}{17{,}100} = 54 \text{ percent of displacement.}$$

Draft = 32 ft.



Load lift = 1000 tons.

Fill one-half the volume as shown.

$$\text{Semicircle } A = \frac{\pi d^2}{8}$$

$$CG = 0.4244\ r.$$

$$\text{Net A} = \frac{\pi}{8}(61^2 - 20^2) = 1460 - 157 = 1300 \text{ ft}^2.$$

$$\text{Water CG} = \overline{R} = \frac{(1460 \times 30.5 - 157 \times 10)\, 0.4244}{1300} = 14 \text{ ft.}$$

T_f is the torque per foot length in ton feet: thus,

$$T_f = \frac{64\, A\overline{R} \cos 45^\circ}{2000} = \frac{64 \times 1300 \times 14 \times 0.707}{2000} = 414 \text{ ft/ton/ft.}$$

Equivalent length = 120 ft for torque.

$$\text{Total cable pull at drum} = \frac{414 \times 120}{31} = 1600 \text{ tons.}$$

Total lift at 6000-ft depth = 1000+ tons.

APPENDIX C. BIBLIOGRAPHY

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